CFD Modelling and Analysis of Different Plate Heat Exchangers

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Abstract. The indirect evaporative cooling system and heat recovery system utilize the return (secondary) air to condition the fresh (primary) air by means of air-to-air heat exchange between the two streams. The temperature difference between the primary and secondary air streams in indirect evaporative cooling system is relatively small. Therefore, efficient heat exchangers should be used since they play a major role in the overall system performance and economics. The parallel plate type heat exchangers have been widely adopted in Indirect Evaporative Cooling (IEC) systems due to their high efficiency in operating at small temperature difference. In this paper we present a theoretical analysis of different designs of counter flow aluminium plate type heat exchanger and results of CFD analysis of pressure drop, flow velocity and thermal effectiveness. For improving the heat transfer between the plates and minimizing the energy loss, the analysis proves useful in the optimization method for selecting parameters of the plate heat exchangers.

Introduction

The current high cost of energy and its impact on the environment are a sufficient motive to reduce the energy consumption of air conditioning systems. Exhaust air heat recovery in air conditioning systems is a technique that widely used to maintain adequate indoor air quality without affecting the quality of the supply air. The maintaining an adequate level of indoor air quality (IAQ), requires an effective energy recovery facilities to treat the ventilation air.

The fixed plate's heat exchangers with its high efficiency (50-80%) are a popular option in the energy recovery ventilation units (ERV), with its high ability to…. sensible and latent heat transfer between the exhaust air and fresh air in separate air paths with relatively large contact areas [1]. Its sensible and latent heat transfer abilities made the ERV system suitable for energy recovery in both heating and cooling processes. Hence the ERV systems have been successfully used in Europe for many years, and become popular in China in recent years [2].

Indirect Evaporative Cooling system (IEC) is an alternative to utilize the energy of the exhaust air, specifically. In an IEC system, parallel vertical plates form the cooling surfaces. While the exhaust air and finely atomised water circulated on one side of the plates by evaporative cooling effect, the dry air is passing on the other side of the plate to be cooled due to sensible heat exchange effect. Using a face part liquid film would result in a higher heat transfer coefficients, increasing the overall mass transfer coefficient and reducing the area required for heat exchange [3-5].

The performance of the fixed-plate heat exchanger has been widely studied due to not only the importance of energy recovery, but its special applications in which the temperature difference in the exhaust air and fresh air is small. Coupled with counter-current flow, the cross flow in a fixed-plate heat exchanger can achieve close end-temperature due to the extended time and surfaces for heat transfer, resulting inefficient heat recovery, [6].

One of the key methods to improve the heat exchanger effectiveness is to increase the heat transfer coefficients between the wall sides. This can be achieved by enhancing the turbulence of the fluids within the channels by means of turbulence promoters or controlling the spacing between the plates. However these options would cause either a pressure loss or increasing the required plate surface area [7].

The performance of a plate heat exchanger (PHE) optimisation by adjusting corrugation pattern on plate surface was revealed that, replacing the flat plate by a Chevron-type plate led to major changes to the velocity vectors behaviour and creates the angular and eddy forms. It was shown that the increased Reynolds number led to the increase of the Nusselt number and the pressure drop while lowering the Fanning friction factor [8-10]. The plate spacing must be adjusted to the value that optimize the heat transfer coefficient and minimize the pressure drop across the heat exchanger [11] The heat transfer coefficient and pressure drop increase with the increase of the air inlet angle [12]. The low plate thickness would result in higher effectiveness, although the material property considered as a secondary effect [13].

This study was aimed to compare the performance of two plate heat exchangers, one with sinusoidal corrugated surface plate and the other pinned surface plate. The quasi-counter-flow arrangement by means of numerical simulation was performed using the SolidWorks Flow Simulation package.

Mathematical Modelling

The quasi-counter-flow plate scheme has been widely used in the HVAC industry recently, because its geometry provides both cross flow and counter flow paths for the air [13]. As shown in Fig. 1, at the entrances, both streams flow in cross-flow arrangement. The pins initiate the turbulent flow before the streams flow through the counter flow region displayed by the straight white line, then re-enter the second cross flow area before leaving the heat exchanger.

As shown in Figure 1, the cold air enters the heat exchanger at the flow inlet (1) at dry-bulb temperature of 290.15 [K], Relative humidity (RH) of 90% and velocity of 1 [m/s]. The hot air leaves the channel at flow outlet (1) to a specified static pressure. The hot air with temperature 300.15 [K] 60% RH and velocity of 1 [m/s] enters the heat exchanger from flow inlet (2). It passes the heat exchanger channels without mixing with the cold stream and leave the heat exchanger from flow outlet (2).

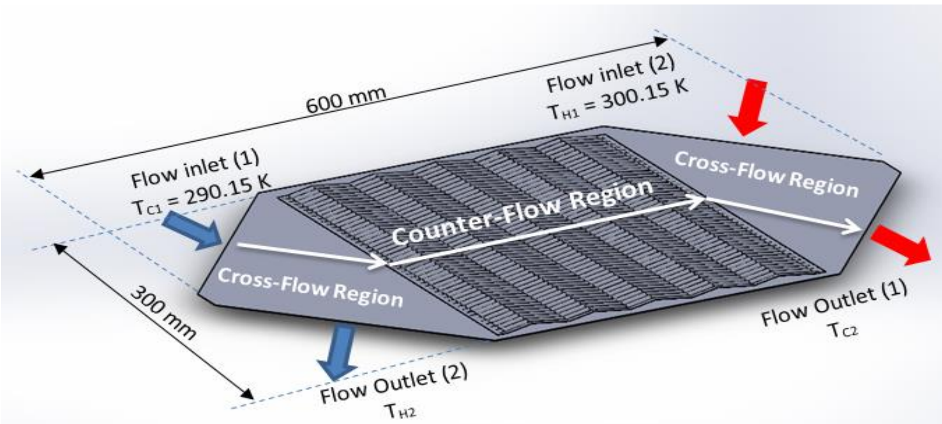

Fig 1. Schematic of Plate heat exchanger model and boundary conditions.

The heat exchanger is treated as a simple air-to-air heat exchanger, with phase changing of water liquid to vapour. The air is assumed as an incompressible gas[13]. The input of the air properties are thermal conductivity (k) = 2.624 10^{-5} [kW/m.k], kinematic viscosity (v) = 17.894 x 10^{-6} [m^2/s] and density (ρ) = 1.225 [kg/m^3].

Table 1 provides the specifications of two types of plate heat exchangers to be investigated. The major difference between these two heat exchangers is that Type 1 has corrugations and Type 2 does not.

Table 1. Heat exchanger specifications.

	Type 1	Type 2
Height of Plate *H [m]*	0.6	0.6
Width of Plate *W [m]*	0.3	0.3
Plate thickness *Δx [m]*	0.0005	0.0005
Gap between two plates *b [m]*	0.0035	0.0035
Number of corrugations	20	0
Corrugation angle	90˚	-
Corrugation width [m]	0.015	-
Corrugation pitch [m]	0.005	-
Effective heat transfer area per plate [m^2]	0.19	0.166
Number to plates *N*	5	5
Number of hot air channels N_h	2	2
Number of cold air channels N_c	2	2
Thermal conductivity (Aluminium) k_{al} [W/m·K]	205.0	205.0

Heat exchanger effectiveness is defined by Eq. (1).

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{\dot{C}_C * (T_{C,1} - T_{C,2})}{\dot{C}_{min} * (T_{C,1} - T_{H,1})} \qquad = \frac{\dot{C}_H * (T_{H,1} - T_{H,2})}{\dot{C}_{min} * (T_{H,1} - T_{C,1})} \tag{1}$$

Where ε denotes the heat exchanger effectiveness. $\dot{Q}$ is the heat transfer between the plate sides [W], $\dot{C}$ is the heat capacity of the flow ($\dot{C} = \dot{m}c_{pa}$). The subscriptions *C* and *H* are for cold (primary) and hot (secondary) flow streams respectively. The minimum $\dot{C}$ ($\dot{C}_{min}$) is the lower capacity rate [$\dot{C}_{min} = \min(\dot{C}_C, \dot{C}_H)$] and is used to calculate the maximum heat exchange across the plate by:

$$\dot{Q}_{max} = \dot{C}_{min} * abs(T_{H,in} - T_{C,in}) \tag{2}$$

In convection heat transfer between the air and the plate within the PHE channel, the heat transfer correlation is expressed in a dimensionless Nusselt number, Nu, the Reynolds number, *Re* and Prandtl number, P_r [13]:

$$Nu = Z.Re^n.Pr^{0.33} = \frac{hD_h}{k_f} \quad , \quad Re = \frac{\rho V D_h}{\mu} \; , \quad Pr = \frac{C_p \mu}{k_f} \tag{3}$$

Where *Z* and *n* are constants depending upon the flow regime and geometrical characteristics of the plates [7]. The hydraulic diameter is given by:

$$\mathrm{D_h} = 4A_{Plate}/P \tag{4}$$

Thermal properties of the fluid are calculated at average outlet temperatures. The overall heat transfer coefficient U is calculated as:

$$U = \frac{Q_{avg}}{A * LMTD} \tag{5}$$

Where *A* is the heat transfer area, Q_{avg} the average heat transfer [$Q_{avg} = (Q_C + Q_H)/2$] and LMTD is the logarithmic mean temperature difference calculated by Eq. (5):

$$LMTD = \frac{(T_{H,1}-T_{C,2})-(T_{H,2}-T_{C,1})}{ln\frac{(T_{H,1}-T_{C,2})}{(T_{H,2}-T_{C,1})}} \quad (6)$$

The overall heat transfer coefficient U can also be calculated from the heat transfer rate from the hot air side to the cold air side across the plate wall as:

$$U = \frac{1}{\left(\frac{1}{h}\right)_C+\left(\frac{\Delta x}{k_{al}}\right)_{Plate}+\left(\frac{1}{h}\right)_H} \quad (7)$$

Eq (7) can be re-arranged to calculate the value of the heat transfer coefficient of the cold side of the plate (h_c):

$$h_C = \left(\frac{1}{U} + \frac{\Delta x}{k_{al}} + \frac{1}{h_H}\right)^{-1} \quad (8)$$

Assuming $h_C = h_H$ due to the equality of flow velocities in the cold and hot channels, the convective heat transfer coefficient at the plate wall is then computed as:

$$h_C = 2 * \left(\frac{1}{U} + \frac{\Delta x}{k_{al}}\right)^{-1} \quad (9)$$

Results and discussion

The CFD code (SolidWorks Flow Simulation) was used to simulate the thermal performance of a parallel plate heat exchanger as described in Table 1 with the inputs described above and *k*- *ε* turbulence model. The *k*- *ε* model is a simple two equations turbulence model suitable for the fully developed turbulence flow. The equations governing the flow of air with the necessary simplifications are obtained from the continuity equation, the equation of momentum, the transport equation of turbulent kinetic energy k and the transport equation of dissipation rate of turbulent kinetic energy ε. The simulation outcomes are listed in Table 2.

Table 2. Simulation Results.

	$T_{H,1}$ [K]	$T_{H,2}$ [K]	$T_{C,1}$ [K]	$T_{C,2}$ [K]	ε	Q_H [W]	Q_C [W]	Q_{avg} [W]	LMTD	h_C
Type 1	300.2	291.16	290.2	297.8	0.765	$9.45x10^{-4}$	$-7.95x10^{-4}$	$7.54x10^{-5}$	1.6	$5x10^{-5}$
Type 2	300.2	292.54	290.2	297.11	0.696	$8.0x10^{-4}$	$-7.2x10^{-4}$	$3.95x10^{-5}$	2.65	$1.9x10^{-5}$

Figure 2 shows that the temperature distribution in the Type 1 heat exchanger with the corrugated shape plate is more homogeneous than that in the Type 2 heat exchanger with the flat plate and turbulence generating pins. This is mainly due to the larger heat transfer area of 0.19 m^2 in Type 1 comparing to 0.166 m^2 in Type 2. The temperature differences in plate sides were reported too small due to small plate thickness.

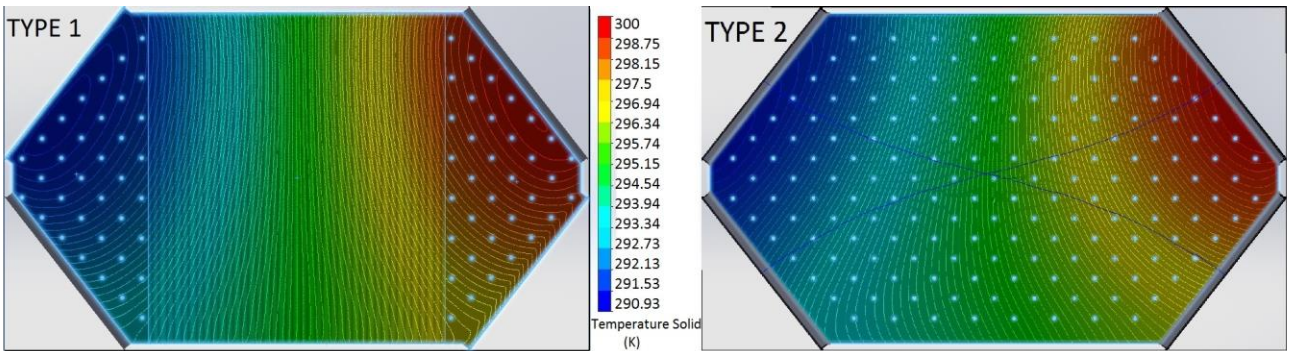

Fig. 2 Temperature distribution over the plate area.

The pressure drop in both cases was 11.02 Pa for type 1 and 10.56 Pa for type 2 (Per channel). Figure 3 shows the relative pressure ditsribution for the air flow over the channel area. The reasonable high pressure drop in the second type is mainly related to the vorticies generated the pins along the flow channels. On the other hand, the pressure drop in the Type 1 could be mainly related to the high turbulence which has been generated at the first corrugation channel (Fig. 4).

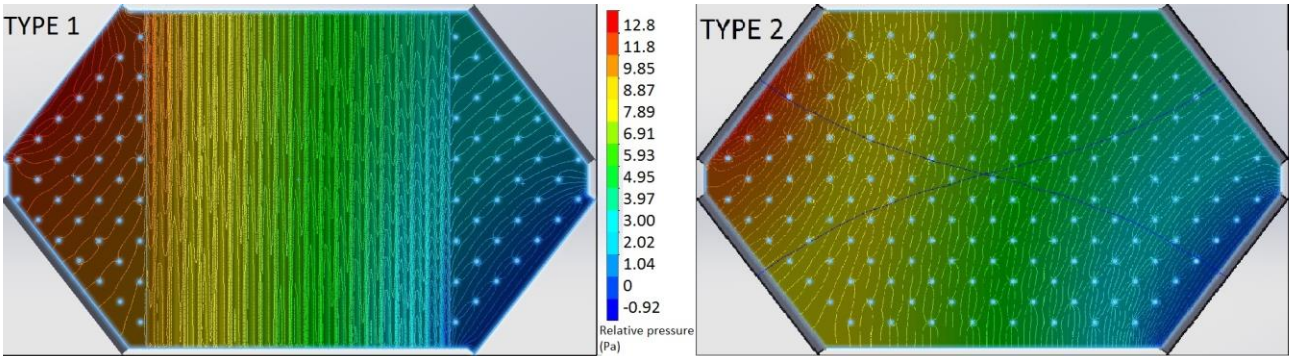

Fig. 3 Relative pressure distribution of fluid over the plate area.

In this study, the effectiveness of the heat exchanger in the case of the corrugated face was found 10% higher than the case of the pins type which is mainly related to the parameters which have been mentioned previously in the introduction (i.e. the channel cross sectional area and Reynolds number).. The sinusoidal surface extended the travel path of air in the counter flow section from 0.35 [m] to 0.395 [m]. On the other side the effect of the turbulence generators (Pins) was found to increase the vorticity in the second case (Fig.4) that resulted in notable elevating of the turbulence level.

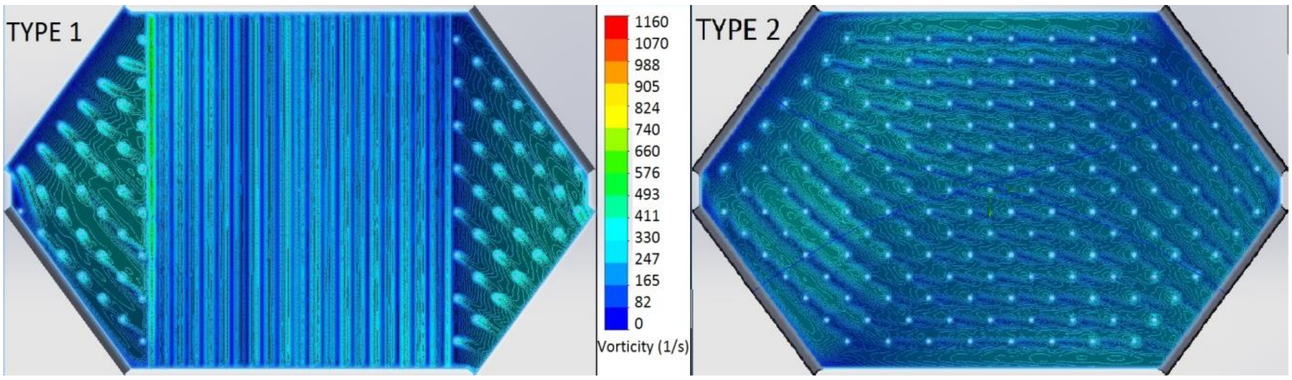

Fig. 4 Fluid vorticity distribution over the plate area.

The numerical simulation showed that the corrugated surface and the high density pinned surface can significantly increase the near wall turbulent mixing level by producing strong vortex flows, which therefore particularly enhances the convective heat transfer in the channel. Compared to the pinned channel, the corrugation enlarges the heat transfer cross section area, which initiated a flow blockage and pressure loss the channels. The cost of pressure drop consists of the fan capital cost and the electricity power consumed by the fan motor [14]. However, enhancement of fluid mixing and formation of recirculation regions improves heat transfer through the plate [9, 15].

Conclusions

Heat transfers in two types of parallel-plates channels in a quasi-counter flow heat exchanger are investigated. Effect of the plate face shape on heat exchange effectiveness is discussed. Conclusions can be drawn as follows:

(1) The whole plate can be divided into three distinct zones: a pure-counter flow zone, and two cross-like flow zones. Most of the areas are these with counter flow arrangement, the least are these with cross flow. Though that the best performance is at the counter flow area, the cross flow area is important to separate the two streams in inlets and outlets.

(2) For the quasi-counter total heat exchanger effectiveness lays between those for pure-counter flow and those for pure cross flow arrangements. The corrugated plate provided more ideal counter flow area rather than the pinned face where the effectiveness was significantly improved by 10%.

As a result, in terms of effectiveness, the corrugated plate heat exchanger is preferable option over the pins plate heat exchanger to employ in indirect evaporative cooling and energy recovery systems. On the other hand, an experimental work must be conducted to investigate the surface wettability of both geometries.

Higher efficiency of heat exchanger can be obtained with greater number of plate’s corrugations. The effect of the pressure drag is increased in the case of the waved plate; the wave amplitude should be chosen to avoid excessive increase of pressure drag. The Reynolds number will increase with corrugated surface over pinned surface, consequently heat transfer coefficient to be raised. A correlation between the wavelength of the corrugation and the friction factor must be investigated further.

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